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INFLUENCE OF COIL DIAMETER AND AXIAL PITCH ON FLOW CHARACTERISTIC OF ADIABATIC HELICAL CAPILLARY TUBE

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ABSTRACT

Capillary tube is an expansion device with small bore used in the refrigeration system, to connect the condenser and evaporator. The capillary tube may be straight or helical or spiral in shape. This paper presents effect of coil diameter and axial pitch of the helical capillary tube on the pressure distribution along it by using the mathematical model on the refrigerant R134a. The model is based on the recommendation from the literature published. Three different equations for friction factor were also examined for this model in this study. The model is validated by comparison made with the experimental data available in the literature for various cases by using three equations of friction factor. A good agreement was obtained with available experimental data with all equations for friction factor. The results show that the predicted length and the pressure distribution along the helical capillary tube are affected by the axial pitch and helix diameter. This mathematical model is enough precise and valuable to provide an effective means in analysis of the capillary tube performance.

KEYWORDS: Helical Capillary Tube, Expansion Device, Pressure Distribution, Homogeneous Flow Model, Coil Diameter, Pitch

NOMENCLATURE

A	Area (m^2)
D	Coil diameter (m)
d	Tube diameter (m)
f	Friction factor
G	Mass flux
h	Enthalpy (J/kg)
K	Contraction factor
L	Length (m)
p	Pitch (m)
P	Pressure (Pa)
Re	Reynolds number
v	Specific volume (m^3/kg)
V	Velocity (m/s)

x	Quality
μ	Viscosity (kg/m-s)
ρ	Density (kg/m^3)
α	Radius of curvature (m)
$\dot{m}$	Mass flow rate (kg/s)

Sub Script

cond	Condenser
evap	Evaporator
fo	Total flow assumed in liquid phase
s	Straight
sp	Single phase
tp	Two phase

INTRODUCTION

The capillary tube is used for domestic and other refrigeration system because of its simplicity and low cost. The capillary tube having no moving parts, like expansion valve and thus it is less expensive.

The expansion process in capillary tube is driven by two major causes; shear stress between the fluid flow and the tube wall and flow acceleration when the liquid turns into the vapor. The refrigerant pressure drops as it passes through the capillary tube is accompanied by reduction in temperature brought about transfer of enthalpy from remaining liquid to provide the enthalpy of evaporation of the flash vapor.

Since the flow behavior inside the capillary tube is complex, many physical models are necessary to predict the characteristic of the refrigerant flow in capillary tube. In the present study, refrigerant flow characteristic inside the capillary tube have been studied to find out recommended empirical correlations for the friction factor with influencing parameter change.

In past decades, flow characteristics of the refrigerant in the capillary tube have been analyzed by various researchers. Work of some of them related to this study is summarized here. Paliwal et. al. [2006] developed flow model in the capillary tube to design and study the performance of the helical capillary tube. Some geometrical parameters like condenser, evaporator pressures, refrigerant flow rate, degree of sub cooling, tube diameter, internal roughness of the tube, pitch and diameter of the helix can affect the length of the capillary tube is shown in their study. They have found that the model predicts with about 10 % accuracy of the experimental data. Wongwises et. al. [2010a and 2010b] studied that the various geometrical parameters of helical capillary tube and different friction factors affect the flow characteristic of alternative refrigerant. They have found that the conventional refrigerant have smaller capillary length than alternative refrigerant and results show that the coil diameter of helical capillary tube affects the length of the capillary tube and the pitch has not significant effect on the length of the capillary tube. Also the model, working with various friction factors shows the good agreement with experimental data and can be used to design the capillary tube with alternative refrigerant.

Sami [2005] has done experimental analysis of the behavior of the capillary tube with alternative refrigerant under various geometrical parameters. Geometrical parameter like length, internal diameter and inlet condition of the capillary tube has influencing the pressure drop across the capillary tube. He also found that the pressure drop with alternative is more than the R-22. Park et. al. [2007] measured performance of the straight and coiled capillary tube by varying tube geometries and coiled shape by experimentation.

They have found that at same operating condition mass flow rate of the coiled capillary tube decreased by 5 to 16 % more than those of the straight capillary tube. The results are compared with the numerical simulation. Mittal et. al. [2009] have developed the homogeneous model including metastable liquid region for adiabatic flow of refrigerant through spiral capillary tube and also investigated the effect of pitch on the spiral capillary tube by the model. The comparison has been made of the R22 and its alternative, and it has been found that flow characteristic is almost similar. Ali Tarred [2008] studied that numerical analysis of for the selection and geometry of the capillary tube and he also founds that comparison of the predicted data shows good agreement with the experimental data and the present model is suitable for the prediction of the geometry of the capillary tube when using alternative refrigerant. Sinpiboon et. al. [2002] developed the mathematical model to study flow characteristic of non-adiabatic capillary tube. The model is categorized into three case of heat exchange process. The model is validated with the experimental data, and found that model is in good agreement with the experimental data.

They suggest that their model can be used to design capillary tube with alternative refrigerant. Fiorelli et. al. [2002a and 2002b] have done experimental validation and analysis for the refrigerant mixture flowing through adiabatic capillary tube and also carried out a study to understand influence of refrigerant mixture, different operating condition and geometrical parameters on the behavior of the capillary tube. Moreover, Main deviation connected with the delay of vaporization is verified experimentally in their study. Wang et. al. [2006] suggested a mathematical model to predict the flow characteristic of flow through adiabatic capillary tube for R22 and its alternative. He found that the predicted results showing about 10 % accuracy with the experimental results. The effect of pitch of the serpentine and helical capillary tube on the performance of the refrigeration system examined by M A Akintunde [2007]. The results show that the pitch of the helical tube has no influence on the performance of the vapor compression refrigeration system. Moreover: with the increment in pitch and the height of the serpentine coil increases system performance. Also the correlations were derived to describe the relationship between straight and coiled capillary tube and between helical and serpentine capillary tube. A A Imran[2009], Meftah et. al.[2006] and Bo et. al. [2003] have developed the model for the alternative refrigerant flowing through adiabatic capillary tube for R22 and its alternative. Meftah et. al. [2006] develops the model for the capillary tube length as a function of the mass flow rate. The results are found in good agreement when they are compared with the experimental results. A A Imran [2009] and Bo e al [2003] have concluded that their model is very valuable to analyze the capillary tube performance. Steady and transient state numerical simulation of capillary tube developed by O G Valldares [2004], working with pure and mixed refrigerant.

The comparison between the experimental and numerical results show that the accuracy of the model developed in his study. Homogeneous and separated flow model developed by Agrawal et. al. [2008a and 2008b] for transcritical carbon dioxide heat pump system. Results show that the model predicts reasonably well with both the correlation for friction factor and the length of the capillary tube is influencing parameter for the system performance. A non adiabatic homogeneous model for carbon dioxide flow capillary tube is presented by Chen et. al. [2005]. They have concluded that the model can be used for both design and performance evolution. Liang et. al. [2001] examined the possibility of applying the equilibrium two-phase drift flux model to simulate the flow of refrigerant in capillary tube. The flow characteristics

for R134a presented in their study. Arun kumar et. al. [2012] presented performance evolution of alternative refrigerant through the helical capillary tube by varying pitch and helix diameter. They have found that the refrigeration system performs better when the atmospheric temperature is 25°C. Bhattacharya et. al. [2008c] have simulated a non- adiabatic capillary tube in trans-critical carbon dioxide heat pump. The effect of various parameters on various performance indicators was investigated by them. Wongwises et. al. [2000] derived a model for separated flow of refrigerant through the capillary tube. The effect of various parameters was investigated by them and some suitable correlations proposed.

MATHEMATICAL MODELING

Mathematical model adopted for this study is presented in this section. The fundamental equations, governing the flow through the capillary tube are the conservations of mass, momentum and energy. In modeling the flow, these equations along with the second law of thermodynamics have been satisfied for both the single phase and two phase regions. Some assumptions are made in this model are as follows:

- Capillary rube is of constant inside diameter and surface roughness.
- Flow through capillary tube is steady, adiabatic and one dimensional.
- Metastable flow phenomenon is neglected.
- Pure refrigerant flowing out of the condenser is either saturated of sub cooled.
- Entrance effects are neglected.
- Homogeneous two phase flow is assumed

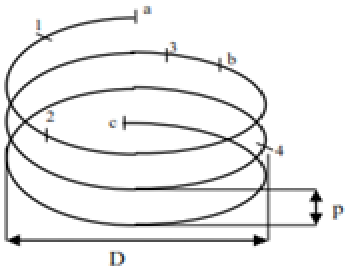

Figure 1: Helical Capillary Tube

The conservation of energy for the steady state adiabatic condition, if the elevation difference is neglected and without external work can be expressed as follows:

$$dh + d\left(\frac{V^2}{2}\right) = 0 \quad (1)$$

Also according to the continuity equation,

$$\dot{m} = \rho A V = constant \quad (2)$$

$$G = \frac{\dot{m}}{A} = \rho V = \frac{V}{v} = constant \quad (3)$$

Where, G is the mass flux.

And the momentum equation:

$$dP + \rho V dV + \frac{\rho V^2}{2}\left(\frac{f dL}{d}\right) = 0 \quad (4)$$

Where, f is the Darcy's friction factor.

Rearranging above equation,

$$-(dP) = \rho V dV + \frac{\rho V^2}{2}\left(\frac{f dL}{d}\right) \quad (5)$$

Pressure drop

$$dP = G dV + \frac{V^2}{2v}\left(\frac{f dL}{d}\right) \quad (6)$$

First term of equation indicated the pressure drop by acceleration and the second one is indicates the pressure drop by friction.

As shown in Figure 1, refrigerant flow can be differentiated in capillary tube in two different regions: the single phase sub cooled liquid region and two phase region.

Single Phase Sub Cooled Liquid Region

In single phase region the refrigerant is in sub cooled liquid region and it is the incompressible fluid. In this region for any point 1 and 2,

$$V_1 = V_2 \quad (7)$$

$$dV = 0 \quad (8)$$

Because of the velocity difference in single phase region is zero. The pressure drop in the single phase region is caused by the virtue of friction only.

The pressure drop in the single phase region is expressed by:

$$dP = \frac{V^2}{2v}\left(\frac{f dL}{d}\right) \quad (9)$$

Rearranging above equation,

$$dL_{sp} = \left(\frac{2vd}{V^2 f_{sp}}\right) dP \quad (10)$$

Where, dL_{sp} is the length of capillary tube in single phase region and f_{sp} is the friction factor for helical capillary tube in single phase region. In the present study three equations for friction factor were used, for first equation,

The transition Reynolds number is increased in a coil and is given as

$$Re_{trans} = 2300\left[1 + 8.6\left(\frac{d}{2\alpha}\right)^{0.45}\right] \quad (11)$$

Where, α is the radius of curvature can be given by

$$\alpha = \frac{D^2 + \left(\frac{p}{\pi}\right)^2}{2D} \tag{12}$$

Where, *p* is the pitch of the helical capillary tube and D is the diameter of helix.

For laminar flow, the ratio of coil to straight pipe friction factor is given by collier [1972],

$$\frac{f_{sp}}{f_s} = \left\{1 - \left[1 - \left(\frac{11\cdot 6}{K}\right)^{0\cdot 45}\right]^{2\cdot 22}\right\}^{-1} \tag{13}$$

Where, $K = Re\left(\frac{d}{2\rho}\right)^{0\cdot 5}$ and *f*s is the friction factor for a straight tube of the same diameter. And for turbulent flow, the ratio of coil to straight pipe friction factor for $Re\left(\frac{d}{2\rho}\right)^2 > 6$ is given by:

$$\frac{f_{sp}}{f_s} = \left[Re\left(\frac{d}{2\rho}\right)^2\right]^{0\cdot 05} \tag{14}$$

To calculate the friction factor *f*s, Churchill's [1977] equation can be used as given below:

$$f_s = 8\left[\left(\frac{8}{Re}\right)^{12} + \frac{1}{(A+B)^{\frac{3}{2}}}\right]^{\frac{1}{12}} \tag{15}$$

Where, $Re = \frac{Gd}{\mu}$

$$A = \left[2\cdot 457 ln\left(\frac{1}{(7/Re)^{0\cdot 9} + 0\cdot 27\frac{\varepsilon}{d}}\right)\right]^{16} \tag{16}$$

$$B = \left(\frac{37530}{Re}\right)^{16} \tag{17}$$

Another equation for friction factor can be used to compute the length is the equation for the friction factor suggested by the Schmidt [1967],

$$\frac{f_{sp}}{f_s} = 1 + 0\cdot 14 Re^x \tag{18}$$

Where,

$$x = \left[1 - \frac{0\cdot 0644}{\left(\frac{D}{d}\right)^{0\cdot 312}}\right] \Big/ \left(\frac{D}{d}\right)^{0\cdot 97} \tag{19}$$

Third equation used to calculate the friction factor, in this model is the friction factor proposed by Mori and Nakayama [1967],

$$f_{sp} = \frac{C_1\left(\frac{d}{D}\right)^{0\cdot 5}}{\left[Re\left(\frac{d}{D}\right)^{2\cdot 5}\right]^{\frac{1}{6}}}\left\{1 + \frac{C_2}{\left[Re\left(\frac{d}{D}\right)^{2\cdot 5}\right]^{\frac{1}{6}}}\right\} \tag{20}$$

$$C_1 = 1 \cdot 88411177 \times 10^{-1} + 85 \cdot 24722168 \left(\frac{\varepsilon}{d}\right) - 4 \cdot 63030629 \times 10^4 \left(\frac{\varepsilon}{d}\right)^2 + 1 \cdot 31570014 \times 10^7 \left(\frac{\varepsilon}{d}\right)^3 \quad (21)$$

$$C_2 = 6 \cdot 79778633 \times 10^{-2} + 25 \cdot 3880380 \left(\frac{\varepsilon}{d}\right) - 1 \cdot 06133140 \times 10^4 \left(\frac{\varepsilon}{d}\right)^2 + 2 \cdot 54555343 \times 10^7 \left(\frac{\varepsilon}{d}\right)^3 \quad (22)$$

Two Phase Region

According to equation (3) for any point 3 and 4,

$$\frac{V_3}{v_3} = \frac{V_4}{v_4} \quad (23)$$

According to equation (1),

$$h_4 + \frac{v_4^2 G^2}{2} = h_3 + \frac{V_3^2}{2} \quad (24)$$

Where, h_4 is given by $h_4 = h_{f4} + x \cdot h_{fg4}$

Substituting the value of h_4 in equation (24) and solving it for *x*, we obtain,

$$x = \frac{(-h_{fg} - v_f v_{fg} G^2) + \sqrt{(h_{fg} + v_f v_{fg} G^2)^2 - 2v_{fg}^2 G^2 A_x}}{v_{fg}^2 G^2} \quad (25)$$

$$A_x = \left(h_f + \frac{v_f^2 G^2}{2} - h_3 - \frac{v_3^2}{2}\right) \quad (26)$$

Using the dryness fraction *x* and the thermodynamics properties, velocity and friction factor at every point determined and used in further calculations.

The pressure drop in the two phase region is occurs due to partly by friction and partly by the acceleration of the refrigerant flowing through the capillary tube.

The pressure drop in the two phase region can be expressed by;

$$dP = GdV + \frac{V^2}{2v}\left(\frac{fdL}{d}\right) \quad (27)$$

Rearranging above equation:

$$dL_{tp,i} = \left(\frac{2vd}{f_{tp,i} V^2}\right)(dP - GdV) \quad (28)$$

Where, $dL_{tp,i}$ is length of the helical capillary tube in two phase region for each section, f_{tp} is the friction factor for the two phase region based on homogeneous region for each section, and dV is the change in velocity along length $dL_{tp,i}$.

Two phase friction factor applicable in the equation (28) can be calculated by the equation proposed by Lin et. al. [1991],

$$f_{tp} = \emptyset_{fo}^2 f_{fo} \frac{v_f}{v_{tp}} \quad (29)$$

Where f_{tp} and f_{fo} are given as,

$$f_{tp} = 8\left[\left(\frac{8}{Re_{tp}}\right)^{12} + \frac{1}{(A_{tp}+B_{tp})^{\frac{3}{2}}}\right]^{\frac{1}{12}} \tag{30}$$

$$f_{fo} = 8\left[\left(\frac{8}{Re_{fo}}\right)^{12} + \frac{1}{(A_{fo}+B_{fo})^{\frac{3}{2}}}\right]^{\frac{1}{12}} \tag{31}$$

Substituting the value of f_{tp} and f_{fo} for equation by replacing Re be Re_{tp} and Re_{fo} defined by

$$Re_{tp} = \frac{Gd}{\mu_{tp}} \tag{32}$$

$$Re_{fo} = \frac{Gd}{\mu_{fo}} \tag{33}$$

Other equations of friction factor used in single phase region can be used in two phase region also; the total length of the helical capillary tube can be expressed as:

$$L_{tp} = \sum_{i=1}^{n} dL_{tp,i} \tag{34}$$

L_{tp} is the length and n is the number of sections of the helical capillary tube considered for the calculation in two phase region.

To calculate length using above equations the MATLAB programming language is used. Finally the total length of the helical capillary tube is obtained by the summation of the length of capillary tube in single phase region and two phase region.

$$L_{Total} = L_{sp} + L_{tp} \tag{35}$$

RESULTS AND DISCUSSIONS

The mathematical model described above is used in the computation of the length of the helical capillary tube and for the analysis of the pressure distribution along the capillary tube with three different equations of friction factor proposed by Collier [1972] and Lin [1991], Schmidt [1967] and Mori and Nakayama [1967]. To validate this model with three different equations of friction factor, the predicted length is compared with the experimental data collected from the other literature for the various cases of vapor compression system with refrigerant R134a. Cases are selected on the basis of different pair of condenser and evaporator pressure and the various pitches, helix diameter, degree of sub cooling, mass flow rate as well as different internal diameter of the helical capillary tube. Table 1 gives the comparison between predicted and experimental results. It could be seen from the Table 1 that the predicted length is near about to the actual length data collected from the experimental data published in the other literature. This model is in reasonable agreement with the experimental results.

Table 1: Comparison between Predicted and Experimental Results

Sr. No.	Diameter (mm)	Pitch (mm)	Helix Diameter (mm)	Condenser Pressure (Bar)	Evaporator Pressure (Bar)	Sub Cooling (°C)	Mass Flow Rate (kg/hr)	Actual Length (m)	Length (m) Predicted by Friction Factor		
									Collier[1972] and Lin[1991]	Mori and Nakayama [1967]	Schmidt [1967]
1.	1.62	10	66	9.1	2.1	1.0	15.12	6.0	6.37	5.67	6.05
2.	1.62	10	66	8.3	1.9	0.8	13.37	6.0	6.91	6.16	6.52
3.	1.62	10	66	6.3	3.6	1.1	10.71	5.5	6.52	5.71	6.06
4.	1.01	6.0	16	10.1	1	2.1	4.30	5.5	6.79	6.04	6.42
5.	1.01	6.0	16	7	1.9	0.9	3.52	5.0	5.10	4.49	4.82

From the above table it is clear that the equations suggested by Schmidt [1967] and Mori and Nakayama [1967] is very nearer to the actual length or the length obtained from the experiment in all cases. Using the equation for friction factor by Collier [1972] and Lin [1991] predicts comparatively more length than the other equations in some cases. Because of the equation by Collier [1972] and Lin [1991] is includes both parameter; pitch and the helix diameter of the helical capillary tube, it can be used to understand the effect of pitch and helix diameter of the helical capillary tube on the pressure distribution. The equations by Schmidt [1967] and Mori and Nakayama [1967] include only the helix diameter as a variable from the geometry of the helical capillary tube, so it can be used to examine the effect of the helix diameter of the helical capillary tube on the pressure distribution.

Figure 2 describes the effect of pitch on length and the pressure distribution of the helical capillary tube and Figure 4, Figure 5 and Figure 6 show that the effect of helix diameter on the pressure distribution and length of the capillary tube, When the P_{cond} is 9.1 bar and P_{eva} 2.1 bar, degree of sub cooling is 1.0°C, mass flow rate of the system is 15.12 kg/hr, diameter of the tube is 1.62 mm.

Figure 3 shows the effect of pitch on length and the pressure distribution of the helical capillary tube and Figure 7, Figure 8 and Figure 9 show that the effect of helix diameter on a pressure distribution and the length of the capillary tube, When the P_{cond} is 8.2 bar and P_{eva} 1.3 bar, degree of sub cooling is 1.4°C, mass flow rate of the system is 4.10 kg/hr, diameter of the tube is 1.01 mm.

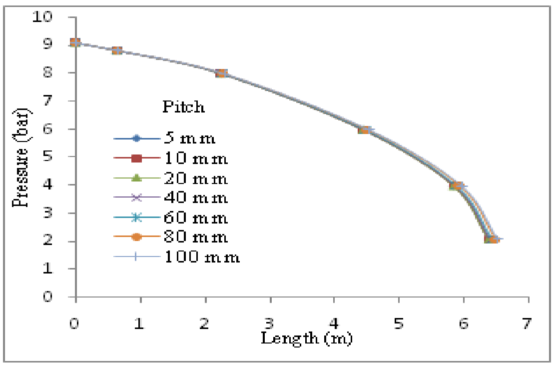

Figure 2: Comparison of Pressure Distribution along Capillary Tube with Larger Value of Coil Diameter

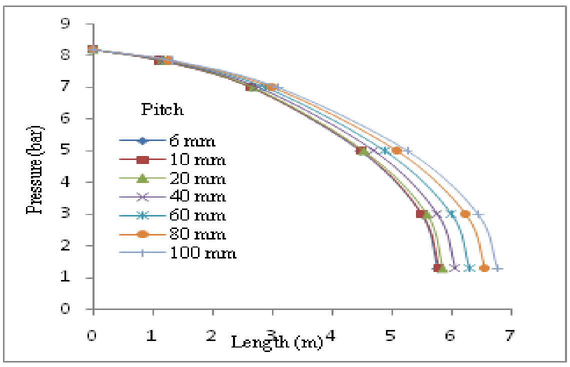

Figure 3: Comparison of Pressure Distribution along Capillary Tube with Smaller Value of Coil Diameter

Effect of pitch on a length and the pressure distribution of the helical capillary tube are shown in Figure 2 and Figure 3, with the use of equation for friction factor proposed by Collier [1972] and Lin [1991], As shown in Figure 2, the pitch is not influencing on the length and the pressure distribution of the helical capillary tube having larger value of helix diameter. Moreover, when the helix diameter has smaller value, the pitch has significant effect on length and pressure distribution of the capillary tube for both the cases of the refrigeration system. Figure 3 shows that the length of the helical capillary tube is increases with the increment in the pitch.

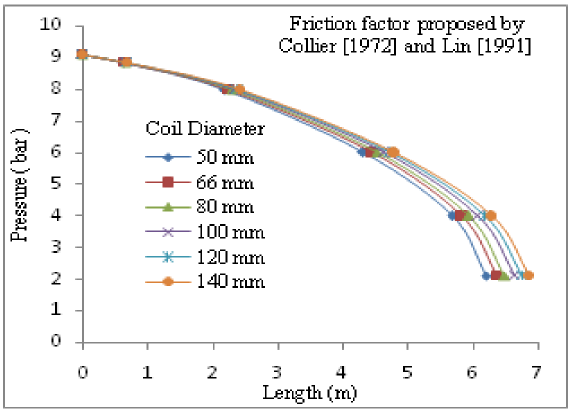

Figure 4: Comparison of Pressure Distribution along Capillary Tube with Larger Value of Pitch

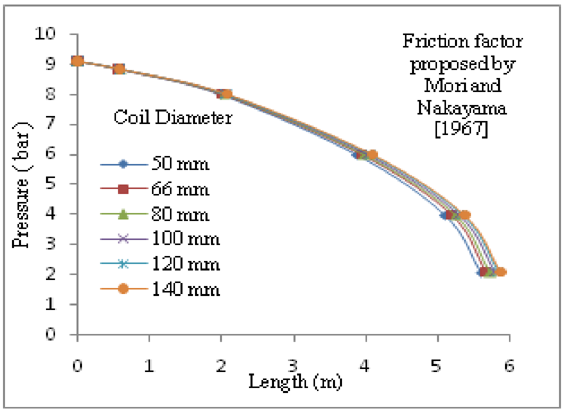

Figure 5: Comparison of Pressure Distribution along Capillary Tube with Larger Value of Pitch

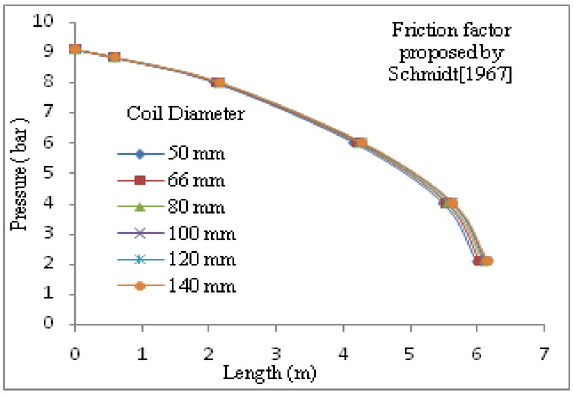

Figure 6: Comparison of Pressure Distribution along Capillary Tube with Larger Value of Pitch

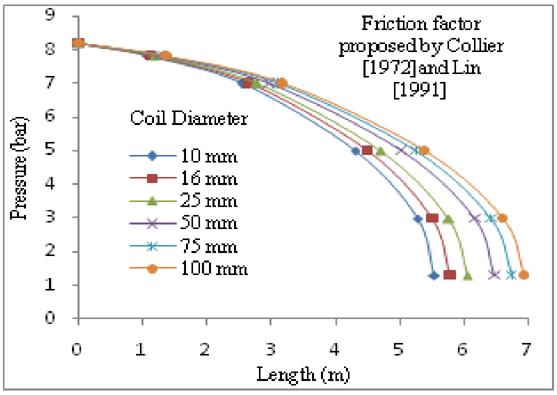

Figure 7: Comparison of Pressure Distribution along Capillary Tube with Smaller Value of Pitch

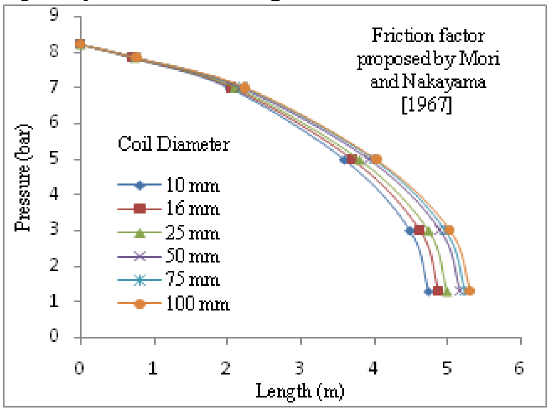

Figure 8: Comparison of Pressure Distribution along Capillary Tube with Smaller Value of Pitch

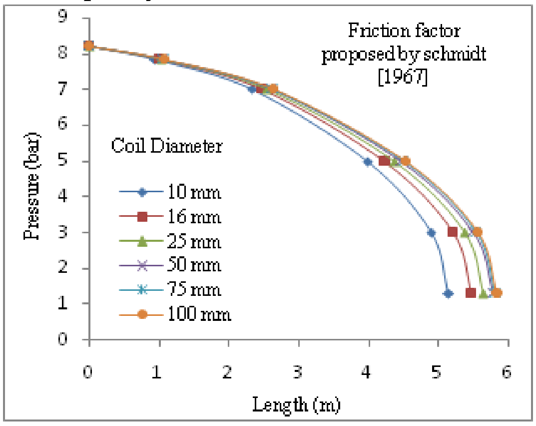

Figure 9: Comparison of Pressure Distribution along Capillary Tube with Smaller Value of Pitch

The effect of helix diameter on the length and pressure distribution has been plotted by using the equation suggested by the Collier [1972], Lin [1991], Schmidt [1967] and Mori and Nakayama [1967], for two cases with the smaller value of pitch and larger value of pitch. Data depicted in the Figures 4 to 9, illustrated that the helix diameter affect significantly on the length and pressure distribution of the helical capillary tube. From the Figures 4, 5 and 6 concluded that, when the pitch of the helical capillary tube has larger value, the length and pressure distribution varies at smaller rate with increase in coil diameter. Data shown in the Figures 7, 8 and 9, demonstrated that the coil diameter influencing more on the length and pressure distribution of the capillary tube when the pitch have smaller value and it also shows that the rate of variation of length and pressure distribution with increment in coil diameter is higher when the capillary tube have

smaller value of axial pitch. The graphs plotted in the figures are in conjunction with published literatures Paliwal et. al. [2006], Wongwises et. al. [2010a and 2010b], Akintunde [2007], Arunkumar et. al. [2012].

CONCLUSIONS

A mathematical model is used to characterize the flow characteristics in adiabatic capillary tube. Conclusion can be made from this study is that the geometries play an expressive role in the design of the capillary tube. The following points can be withdrawn from this study. Equation of friction factor proposed by Mori and Nakayama [1967] is found most suitable for this mathematical model. The pitch is not influencing more on the length and pressure distribution at larger value of coil diameter. Some variation has been found in the predicted length and pressure distribution with the change in pitch, when the helix diameter is smaller. The coil diameter affects significantly to the length and pressure distribution. Effect of coil diameter on the length and pressure distribution is quite sluggish when the pitch has larger value. The numerical model presented in this investigation provides a technical tool for estimation of the capillary tube geometry with acceptable accuracy.

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